CALCULATING FOR SYSTEM EFFICIENCY SOLVES FAN LOSSES

W. David Bevirt, P.E., CMS
NEBB
(520) 825-1049

INTRODUCTION

Heating, ventilating and air conditioning (HVAC) duct system designers are faced with many decisions once load calculations have been completed and the type of HVAC system (constant volume, variable air volume, etc.) has been selected. The following items affect the supply air fan volume and total pressure capacity and therefore the fan selection:

S Duct construction materials

S Duct system layout

S Duct systemCfittings and straight duct

S Duct leakage

S ASystem Effect@

S Testing, adjusting and balancing (TAB)

When a return air fan is included in the system design, this discussion applies to both fans as a unit. The important thing to remember is the *fan affinity laws*. When the fan airflow (L/s or cfm) or the fan speed (rpm) changes only ten percent (10%) the fan static pressure or total pressure changes about twenty-one percent (21%), and the fan power (watts or brake horsepower) changes about thirty-three percent (33%). Therefore, to prevent the TAB contractor from being required to increase fan speeds at the end of the HVAC system installation, the duct system designer must be aware of how the above listed items affect system efficiency.

DUCT CONSTRUCTION MATERIALS

The duct friction loss charts in Chapter 32 of the 1997 ASHRAE Fundamentals Handbook are based on *standard air* flowing through average, clean, round, galvanized metal ducts with beaded slip couplings on 1200 mm (48 inch) centers and an absolute roughness of 0.09 mm (0.0003 ft). The previous duct friction loss charts were based on 750 mm (30 inch) joints and an absolute roughness of 0.15 mm (0.0005 ft). Many computer software programs and duct calculators still use the obsolete older values.

Actual air volume is used to determine duct friction loss using the friction loss charts. This loss is multiplied by correction factor(s) to obtain the adjusted duct friction loss. Correction factors for flexible duct and acoustic duct lining at duct velocities of 10 m/s (2000 fpm) can be as high as 2.34. Therefore a calculated system straight duct static pressure loss of 500 pascals (2 in.w.g.) Would increase to 1170 Pa (4.6 in.w.g.).

DUCT SYSTEM LAYOUT

Many duct system layouts are simple and straight forward with a minimum of duct fittings. Many long and complicated systems with reasonable pressure drops are designed with low pressure loss duct fittings and low aspect ratio rectangular or round spiral ductwork. The problem arises when the system installer has to add additional fittings to avoid beams, piping or conduct of other trades. A few added high pressure loss fittings can double the designer=s calculated total system pressure loss. For example, only one Aaround-the-beam@ fitting (Figure 1), if added without turning vanes where the duct velocity is 10 m/s (2000 fpm), would add 554 Pa (2.31 in.w.g.) to the system pressure loss. The fitting pressure loss is obtained by multiplying the loss coefficient (C) by the system velocity pressure at that point.

METRIC (SI) CALCULATIONS

Velocity pressure of 10 m/s = 60 Pa
Fitting pressure loss (L/H = 2) = 9.24 x 60 = 554 Pa

ENGLISH (IP) CALCULATIONS

Velocity Pressure of 2000 fpm = 0.25 in.w.g.
Fitting pressure loss (L/H = 2) = 9.24 x 0.25 = 2.31 in.w.g.

FIGURE 1 RECTANGULAR DUCT WITH 4-90E MITERED ELLS TO AVOID AN OBSTRUCTION

DUCT SYSTEM COMPONENTS

STRAIGHT DUCT LOSSES

Pressure drop in a straight duct section is caused by surface friction, and varies with the velocity, the duct size and length, and the interior surface roughness. Friction loss for round ducts is determined from Air Duct Friction Loss Charts found in chapter 32 of the 1997 ASHRAE Fundamentals Handbook. They are based on standard air with a density of 1.204 kg/m^3 (0.075 lb/cu. ft.) flowing through average clean round galvanized metal ducts.

In HVAC work, the values from the friction loss charts may be used without correction for temperatures between 10EC to 60EC (50EF to 140EF) and up to 600 m (2000 ft.) altitude. Tables 1 (Page 4) and 2 (Page 5)may be used where air density is a significant factor, such as at higher altitudes or where high temperature air is being handled, to correct for temperature and/or altitude. The actual air volume (L/s or cfm) is used to find the duct friction loss, which is multiplied by the correction factor(s) to obtain the adjusted duct friction loss.

As HVAC duct systems are sized first as round ducts, rectangular duct sizes are selected to provide flow rates equivalent to those of the round ducts originally selected. Tables In ASHRAE Chapter 32 give the circular equivalents of rectangular ducts for equal friction and airflow rates for aspect ratios not greater than 8:1, although ratios above 4:1 are not recommended. Note that the mean velocity in a rectangular duct will be less than the velocity for its circular equivalent. Therefore round or square ducts will be more energy efficient than rectangular ducts. Efficiency decreases as the aspect ratio increases. This also applies to flat oval ductwork.

TABLE 1 AIR DENSITY CORRECTION FACTORS (METRIC UNITS)

Altitude (m)	Sea Level	250	500	750	1000	1250	1500	1750	2000	2500	3000
Barometer (kPa)	101.3	98.3	96.3	93.2	90.2	88.2	85.1	83.1	80.0	76.0	71.9
Air Temp 0E	1.081.0	1.05	1.02	0.99	0.96	0.93	0.91	0.88	0.86	0.81	0.76
EC 20E	0	0.97	0.95	0.92	0.89	0.87	0.84	0.82	0.79	0.75	0.71
50E	0.91	0.89	0.86	0.84	0.81	0.79	0.77	0.75	0.72	0.68	0.64
75E	0.85	0.82	0.80	0.78	0.75	0.73	0.71	0.69	0.67	0.63	0.60
100E	0.790.7	0.77	0.75	0.72	0.70	0.68	0.66	0.65	0.63	0.59	0.56
125E	4	0.72	0.70	0.68	0.66	0.64	0.62	0.60	0.59	0.55	0.52
150E	0.70	0.68	0.66	0.64	0.62	0.60	0.59	0.57	0.55	0.52	0.49
175E	0.66	0.64	0.62	0.62	0.59	0.57	0.55	0.54	0.52	0.440.	0.46
200E	0.62	0.61	0.59	0.57	0.56	0.54	0.52	0.51	0.49	47	0.44
225E	0.59	0.58	0.56	0.54	0.53	0.51	0.50	0.48	0.47	0.44	0.42
250E	0.56	0.55	0.53	0.52	0.50	0.49	0.47	0.46	0.45	0.42	0.40
275E	0.54	0.52	0.51	0.49	0.48	0.47	0.45	0.44	0.43	0.40	0.38
300E	0.51	0.50	0.49	0.47	0.46	0.45	0.43	0.42	0.41	0.38	0.36
325E	0.49	0.48	0.47	0.45	0.44	0.43	0.41	0.40	0.39	0.37	0.35
350E	0.47	0.46	0.45	0.43	0.42	0.41	0.40	0.39	0.38	0.35	0.33
375E	0.46	0.44	0.43	0.42	0.41	0.39	0.38	0.37	0.36	0.34	0.32
400E	0.44	0.43	0.41	0.40	0.39	0.38	0.37	0.36	0.35	0.33	0.31
425E	0.42	0.41	0.40	0.39	0.38	0.37	0.35	0.34	0.33	0.32	0.30
450E	0.41	0.40	0.38	0.37	0.36	0.35	0.34	0.33	0.32	0.31	0.29
475E	0.390.3	0.38	0.37	0.36	0.35	0.34	0.33	0.32	0.31	0.29	0.28
500E	8	0.37	0.36	0.35	0.34	0.33	0.32	0.31	0.30	0.28	0.27
525E	0.37	0.36	0.35	0.34	0.33	0.32	0.31	0.30	0.29	0.27	0.26
Standard Air Density, Sea Level, 20EC = 1.2041 kg/m^3 at 101.325 kPa											

DUCT FITTING LOSSES

As demonstrated above, the actual fitting pressure loss is equal to the loss coefficient (C) times the velocity pressure. Therefore, the most efficient duct fittings have the lowest loss coefficients. Also as the duct velocity pressure increases as the square of the velocity, restraining duct velocities also adds to system fan energy efficiencies.

TURNING VANES MISSING

For many years, sheet metal contractors, often with the system designer's approval, have eliminated every other turning vane from the vane runners installed in rectangular mitered duct elbows. Some contractors even believed that they would lower the pressure loss of duct elbows by doing this. But they were wrong! This practice more than doubles elbow pressure losses, and definitely is not recommended.

TABLE 2 AIR DENSITY CORRECTION FACTORS (I.P. UNITS)

Altitude (ft)	Sea Level	1000	2000	3000	4000	5000	6000	7000	8000	9000	10,000
Barometer (in.Hg)	29.92	28.86	27.82	26.82	25.84	24.90	23.98	23.09	22.22	21.39	20.58
(in.w.g.)	407.5	392.8	378.6	365.0	351.7	338.9	326.4	314.3	302.1	291.1	280.1
Air Temp	1.26	1.22	1.17	1.13	1.09	1.05	1.01	0.97	0.93	0.90	0.87
EF	1.15	1.11	1.07	1.03	0.99	0.95	0.91	0.89	0.85	0.82	0.79
40E	1.06	1.02	0.99	0.95	0.92	0.88	0.85	0.82	0.79	0.76	0.730.69
70E	1.00	0.96	0.93	0.89	0.86	0.83	0.80	0.77	0.74	0.71	0.65
100E	0.95	0.92	0.88	0.85	0.81	0.78	0.75	0.73	0.70	0.68	0.60
150E	0.87	0.84	0.81	0.78	0.75	0.72	0.69	0.67	0.65	0.62	0.55
200E	0.80	0.77	0.74	0.71	0.69	0.66	0.64	0.62	0.60	0.57	0.51
250E	0.75	0.72	0.70	0.67	0.64	0.62	0.60	0.58	0.56	0.58	0.48
300E	0.70	0.67	0.65	0.62	0.60	0.58	0.56	0.54	0.52	0.50	0.45
350E	0.65	0.62	0.60	0.58	0.56	0.54	0.52	0.51	0.49	0.47	0.42
400E	0.62	0.60	0.57	0.55	0.53	0.51	0.49	0.48	0.46	0.44	0.40
450E	0.58	0.56	0.54	0.52	0.50	0.48	0.46	0.45	0.43	0.42	0.38
500E	0.55	0.53	0.51	0.49	0.47	0.45	0.44	0.43	0.41	0.39	0.36
550E	0.53	0.51	0.49	0.47	0.45	0.44	0.42	0.41	0.39	0.38	0.34
600E	0.50	0.48	0.46	0.45	0.43	0.41	0.40	0.39	0.37	0.35	0.32
700E	0.46	0.44	0.43	0.41	0.39	0.38	0.37	0.35	0.34	0.33	0.29
800E	0.42	0.40	0.39	0.37	0.36	0.35	0.33	0.32	0.31	0.30	0.27
900E	0.39	0.37	0.36	0.35	0.33	0.32	0.31	0.30	0.29	0.28	0.25
1000E	0.36	0.35	0.33	0.32	0.31	0.30	0.29	0.28	0.27	0.26	
Standard Air Density, Sea Level, 70EF = 0.075 lb/cu ft at 29.92 in. Hg											

Figure 2 is a chart developed from research performed by ETL Laboratories in Cortland, New York. ETL tested single thickness turning vanes with a radius of 114 mm (4**2** in.). The distance between vanes was varied from 75 mm to 165 mm (3 in. To 6**2** in.) in increments of 6 mm (1/4 in.) using embossed rail runners. Airflow velocities varied from 5 to 12.5 m/s (1,000 to 2,500 fpm) in a 600 mm x 600 mm (24 in. X 24 in.) mitered elbow. The loss coefficient of 0.18 for the standard spacing of 88 mm (3 1/4 in.) may be compared with the loss coefficient of 0.46 at 165 mm (6 **2** in.) spacing (every other vane missing). The pressure loss of the elbow with missing turning vanes was over 2**2** times the pressure loss of a properly fabricated elbow containing all of the vanes.

BRANCH LOSSES

The loss in a branch connection (tee or wye) depends on the ratio of the velocity of the diverted flow to the total flow, the areas of the main duct and branch, and the takeoff geometry. The total pressure loss coefficients for a variety of branch configurations for rectangular ductwork are shown in Figure 3.

FIGURE 2 TURNING VANE RESEARCH

The total pressure loss of a tee or wye is a function of the branch velocity to the upstream main duct velocity using the nomenclature (V_b/V_c) shown in the figures.

For example, data from the duct fitting research program shows that an inexpensive 45E entry branch from a rectangular main (Table A) is a far more efficient fitting to use than a rectangular branch with an expensive extractor (Table D). Using a V_b/V_c velocity ratio of 1.0, and a Q_b/Q_c airflow ratio of 0.5, the following can be extracted from the tables and compared.

If a commonly used round branch (Table B) or rectangular branch (Table C) is added to the fitting loss coefficient comparison, one can see that the use of extractors should be eliminated, as they also can create other problems immediately downstream in the main duct. They basically barely improve the fitting loss coefficient value.

FIGURE 3 FITTING LOSS COEFFICIENTS

To compare the loss coefficient value under the same conditions:

S 45E entry branch, C = 0.74

S round branch, C = 1.26

S rectangular branch, C = 1.27

S extractor in branch, C = 1.21

The 45E entry fitting is 164% more efficient than the extractor fitting.

Regardless of the configuration used, all branches should contain a balancing damper.

DUCT AIR LEAKAGE

The amount of air leaking from duct systems is not a mystery. Table 3 (Page 10), Table 4 (Page 11), and Figures 4 and 5 (Page 9) may be used by HVAC system designers to predict the amount of air leakage based on their duct plans and specifications. The calculated amount of air leakage should be added to the selected fan capacity and so noted either in the specifications or on the drawings. This allows TAB technicians to more easily verify actual system conditions, and properly specified sizes of adjustable fan drives should be able to handle any changes required by the TAB work. Many fan drives, fan motors, and electrical services to the fans have had to be changed because duct system air leakage was ignored by the system designer.

The *leakage classes* in Table 3 (Page 10) are based on seal classes (how the ductwork is sealed with mastic). The leakage class designation is based on the average amount of leakage (L/s per square meter or cfm per 100 square feet) of duct surface with an average internal static pressure of 250 pascals (1 inch water gauge). The charts in Figures 4 and 5 (Page 9) are used for other average internal duct pressures.

METRIC EXAMPLE

In metric units, 465 square meters of rectangular duct at 250 pascals average pressure using seal class C and leakage class 24 (transverse joints only) would leak 558 L/s (465 x 1.2 = 558 L/s).

Using seal class B and leakage class 12, the same ductwork would leak 279 L/s (465 x 0.6 = 279 L/s). Using seal class A, the same ductwork would leak 140 L/s (465 x 0.3 = 140 L/s) using leakage class 6. The ductwork in each case must be constructed in the required pressure classification metal gauges and related reinforcement.

FIGURE 4 DUCT LEAKAGE CLASSIFICATIONS (METRIC)

FIGURE 5 DUCT LEAKAGE CLASSIFICATIONS (I.P.)

I.P. EXAMPLE

5000 square feet of rectangular duct at 1 in.w.g. average pressure using seal class C and leakage class 24 (transverse joints only) would leak 1200 cfm (5000 x 24/100 = 1200 cfm).

Using seal class B and leakage class 12, the same ductwork would leak 600 cfm (5000 x 12/100 = 600 cfm). Using seal class A, the same ductwork would leak 300 cfm (5000 x 6/100 = 300 cfm) using leakage class 6. The ductwork in each case must be constructed in the required pressure classification metal gauges and related reinforcement.

PREDICTING LEAKAGE

Table 4 allows a system designer to predict the percent of duct air leakage by dividing the HVAC unit fan capacity by the total square meters (square feet) of duct surface, using the leakage class, and then the average internal duct static pressure. For example, if the above 465 m^2 (5000 ft^2) duct system had a 6975 L/s (15,000 cfm) fan, the L/s per m^2 in the second column of Table 4 would be 15 (6975/465 = 15) or in I.P. units, the cfm/ft^2 in the second column of Table 4 would be 3 (15,000/5,000 = 3). A duct system with a leakage class of 6 (seal class A) would have 4.1 percent air leakage at 750 Pa (3 in.w.g.) average or 3.1 percent leakage at 500 Pa (2 in.w.g.) average. Table 4 does not include HVAC unit or terminal unit connection leakage, which must be added.

TABLE 3 APPLICABLE LEAKAGE CLASSES

DUCT CLASS	1/2, 1, 2 in.w.g. (125, 250,500 Pa)		3 in.w.g. (750 Pa)	4, 6, 10 in.w.g. (1000,1500,2500 Pa)
SEAL CLASS	NONE	C	B	A
APPLICABLE SEALING	N/A	TRANSVERSE JOINTS ONLY	TRANSVERSE JOINTS AND SEAMS	ALL JOINTS, SEAMS AND WALL PENETRATIONS
LEAKAGE CLASS (C_L) cfm/100 sq. Ft (L/s per m^2) at 1 in.w.g. (250 Pa0				
RECTANGULAR METAL	48	24	12	6
ROUND AND OVAL	30	12	6	3
ROUND AND OVAL METAL	N/A	6	N/A	N/A
ROUND FIBROUS GLASS	N/A	3	N/A	N/A

The often specified "maximum of 1% leakage" is almost impossible to attain under normal system design conditions (one percent leakage or less are the shaded numbers in Table 4). Two to ten percent duct air leakage generally is found throughout the industry for average size HVAC systems. Note in Table 4 that the larger the system and/or the higher the average duct static pressure, the greater the duct leakage will be, even in the best, totally sealed duct systems.

TABLE 4 LEAKAGE AS A PERCENTAGE OF SYSTEM AIRFLOW

LEAKAGE CLASS	System Airflow cfm/ft²	L/s per m²	AVERAGE STATIC PRESSURE in.w.g. (Pa) 1/2 (125)	1 (250)	2 (500)	3 (750)	4 (1000)	6 (1500)
48	2	10	15	24	38			
	2.5	12.7	12	19	30			
	3	15	10	16	25			
	4	20	7.7	12	19			
	5	25	6.1	9.6	15			
24	2	10	7.7	12	19			
	2.5	12.7	6.1	9.6	15			
	3	15	5.1	8.0	13			
	4	20	3.8	6.0	9.4			
	5	25	3.1	4.8	7.5			
12	2	10	3.8	6	9.4	12		
	2.5	12.7	3.1	4.8	7.5	9.8		
	3	15	2.6	4.0	6.3	8.2		
	4	20	1.9	3.0	4.7	6.1		
	5	25	1.5	2.4	3.8	4.9		
6	2	10	1.9	3	4.7	6.1	7.4	9.6
	2.5	12.7	1.5	2.4	3.8	4.9	5.9	7.7
	3	15	1.3	2.0	3.1	4.1	4.9	6.4
	4	20	**1.0**	1.5	2.4	3.1	3.7	4.8
	5	25	**0.8**	1.2	1.9	2.4	3.0	3.8
3	2	10	**1.0**	1.5	2.4	3.1	3.7	4.8
	2.5	12.7	**0.8**	1.2	1.9	2.4	3.0	3.8
	3	15	**0.6**	**1.0**	1.6	2.0	2.5	3.2
	4	20	**0.5**	**0.8**	1.3	1.6	2.0	2.6
	5	25	**0.4**	**0.6**	**0.9**	1.2	1.5	1.9

Using the fan laws, a 4.1 percent air leakage will increase 7.5 watts fan power (10 fan brake horsepower) to 8.46 W (11.28 BHP).

$$\frac{FP_2}{FP_1} = \left(\frac{Q_2}{Q_1}\right)^3 ; \; FP_2 = 7.5 \left(\frac{1.041}{1.0}\right)^3 = 8.46 \text{ W} \quad \text{or}$$

$$FP_2 = 10 \left(\frac{1.041}{1.01}\right)^3 = 11.28 \text{ BHP}$$

So, 4.1 percent leakage causes over an eleven percent increase in the fan power requirements. A more realistic 6.1 percent leakage would increase the 7.5 W (10 BHP) requirements to 8.96 W (11.94 BHP), over a nineteen percent increase.

$$FP_2 = 7.5 \left(\frac{1.061}{1.0}\right)^3 = 8.96 \text{ W}$$

$$FP_2 = 10 \left(\frac{1.061}{1.0}\right)^3 = 11.94 \text{ BHP}$$

Unfortunately, many system designers do not include this extra fan airflow and energy requirements in their HVAC unit fan selection. However, TAB technicians usually can determine the amount of duct air leakage using field measurements.

SYSTEM EFFECT

System Effect is the derating or loss of capacity of a fan caused by poorly designed duct fittings at, or close to, the fan discharge and inlet. In addition to the generally unknown problem of system effect, **TAB technicians cannot measure system effect in the field.** Approximate fan capacity losses caused by system effect only can be calculated using dimensional measurements of the fan-ductwork connections and using data from tables and charts found in the Air Movement and Control Association (AMCA) Publication 201**C**Fans *and Systems.* This information also has been reprinted in ASHRAE, NEBB and SMACNA publications.

The diagonal lines in Figure 6 (Page 13) are called *system effect curves.* Any deviation from a straight piece of fan discharge duct within the "effective length" distance shown in Figure 7 (Page 13) may create a *system effect. The* reason is that centrifugal fans are tested and rated by AMCA exactly as shown in Figure 7, i.e., no inlet duct and a straight discharge duct of at least the length of 2-1/2 duct diameters.

FIGURE 6 SYSTEM EFFECT CURVES (AMCA)

FIGURE 7 SYSTEM EFFECT CURVES FOR OUTLET DUCTS (AMCA))

For example, an exhaust fan on a roof without a discharge duct (assume a blast area ratio of 0.7 in Figure 7) would have an AS@

system effect curve. Using Figure 6 [assume an outlet velocity of 10 m/s (2000 fpm)], a pressure loss of 50 Pa (0.2 in.w.g.) is obtained. That means that the rated static pressure capacity of the fan is lowered by 50 Pa (0.2 in.w.g.).

FIGURE 8 OUTLET ELBOWS ON SWSI CENTRIFUGAL FANS (AMCA)

If an elbow (position C) was installed directly at the fan discharge, a "P" system effect curve would be obtained from Figure 8. Using the same 10 m/s (2000 fpm) in Figure 6 (Page 13), the system effect loss is 125 Pa (0.5 in.w.g.).

System effect losses also must be calculated for most fan inlet duct connections. Combined losses for both the inlet and outlet of a fan easily can exceed 250 Pa (1 in.w.g.) for each HVAC system fan. That could be a system total of 500 Pa (2 in.w.g.) when both a supply air fan and return air fan are used.

Figure 9 illustrates deficient fan/system performance caused by system effect. Curve A shows the HVAC system airflow capacity and pressure loss that has been calculated. The system designer selected the fan to operate at Point 1 on system curve A. However, no allowance has been made for the effect of duct system connections on the fan performance. To compensate, a system effect factor must be added to the calculated system pressure losses to determine the new system curve B that should be used to select the fan.

FIGURE 9 EFFECTS OF SYSTEM EFFECT (ASHRAE)

The point of intersection between the initially selected fan performance curve and this new "phantom" system curve B is Point 4. Therefore, the actual system flow volume is deficient by the difference from Point 1 to Point 4. Often the fan manufacturer will be blamed for deficient fan performance!

To achieve the design airflow volume, a system effect factor equal to the pressure difference between Point 1 and point 2 must be added to the calculated system pressure losses. The fan should be selected to operate at point 2 where a new, higher fan rpm curve crosses phantom system curve B. A higher fan power (watts or brake horsepower) also will be required.

When a TAB technician measures the actual HVAC system conditions with the corrected fan rpm, the airflow volume and static pressure will be established as point 1, because that is where the system actually is operating. The system is not operating on the phantom system curve, which was used only to select the derated capacity fan rpm. System effect *cannot* be measured in the field, but only calculated after a visual inspection is made of the fan/duct system connections.

Because system effect is velocity related, the difference between points 1 and 2 is greater than the difference between points 3 and 4. The system effect factor includes only the effect of the system configuration on the fan's performance. All duct and duct fitting pressure losses are calculated as part of the HVAC system pressure losses and remain a part of *system curve A*.

TESTING, ADJUSTING AND BALANCING

All of the above possible problems should be found by a TAB technician employed by a nationally certified TAB firm when the system is commissioned. However, corrections at that time can be very costly. Taking the worst case scenarios from the above system capacity loss problems, if they were applied to more than one HVAC system in a building, they could present a monumental task to the TAB technicians and all others involved.

EXAMPLE

A 6975 L/s (15,000 cfm) airflow system operating at 971 Pa (3.9 in.w.g.) With 465 m^2 (5000 ft^2) of rectangular metal duct surface will be used for this example. It is assumed that no allowances have been made by the design engineer.

First of all, if the sheet metal contractor installed internally lined ductwork instead of externally insulating it, an increase of 670 pascals (2.6 in.w.g.) Of static pressure would be added to the fan capacity.

Next, if the around-the-beam fitting was installed as described with turning vanes missing, another 554 pascals (2.31 in.w.g.) would be added.

For duct leakage, Table 4 (Page 11) indicates a 3.1 percent leakage at a 500 Pa (2 in.w.g.) average duct pressure (leakage class 6). To calculate the losses:

6975 L/s x 0.031 = 216 L/s loss

15,000 cfm x 0.31 = 465 cfm loss

Then add approximately one percent for HVAC unit and terminal unit connection losses. The duct system leakage totals are:

216 + 70 = 286 L/s

465 + 150 = 615 cfm

With a 90E elbow directly attached to the HVAC unit looking up (position C in Figure 8 on Page 14), the system effect static pressure (SP) loss is 125 Pa (0.5 in.w.g.). Factory-built HVAC units often have system effect on the inlet side of the fan(s) that usually does not exceed 63 Pa (0.25 in.w.g.), which is used for this example.

Fan inlet SP loss	=	63 Pa (0.25 in.w.g.)
Fan outlet SP loss	=	125 Pa (0.50 in.w.g.)
Total fan system effect	=	188 Pa (0.75 in.w.g.)

To total all of the above increases to the specified fan capacity:

Item	**AirflowCL/s (cfm)**	**SPCPa (in.w.g.)**
Specified fan	6975 (15,000)	971 (3.90)
Straight duct loss	- - -	670 (2.60)
Bream fitting loss	- - -	554 (2.31
Duct air leakage	286 (615)	- - -
System Effect	- - -	188 (0.75)
Revised fan capacity	7261 (15,615)	2383 (9.56)

Using a table from a typical fan catalog, one finds that a forward curved (FC) fan producing 7261 L/s at 2388 Pa (15,615 cfm at 9.56 in.w.g.) SP requires 28.7 W (38.2 BHP) and is Class II constructed fan at 1500 rpm.

The specified 6975 L/s (15,000 cfm) fan requires 9.4 W (12.5 BHP) and is a Class I constructed fan operating at 1143 rpm. The duct system connected to the HVAC unit now is actually handling 7261 L/s at 2195 Pa (15,615 cfm at 8.81 in.w.g.), which puts it into a much heavier 2500 Pa (10 in.w.g.) construction pressure classification. [System effect of 188 Pa (0.75 in.w.g.) is not included for duct losses]. As the average duct system pressure is higher, duct leakage also must be recalculated. The ductwork is now heavier, but still in leakage class 6, so the leakage is close to 5 percent, increasing the total fan airflow to 7324 L/s (15,750 cfm).

Finally, the only way to avoid the above problems is to 1) employ a competent HVAC system design engineer who realistically includes system leakage and proper HVAC unit connections and duct fittings in the HVAC system plans and specifications; 2) inspect the system components and installation methods used during construction; and 3) use a certified testing, adjusting and balancing (TAB) firm to monitor the above work from the bidding stage to the completion of construction. Then there will be no surprises, no problems, and no delays to keep from putting often costly HVAC systems into operation. Also the HVAC System owner would not be paying over 3 times the ongoing cost of electrical energy for each system with similar problems for years to follow.

AUTHOR=S ADDRESS:

W. David Bevirt, P.E., CMS
NEBB
14046 North Fawnbrooke Drive
Tucson, Arizona 85737